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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

The Nature of the Lubrication Process

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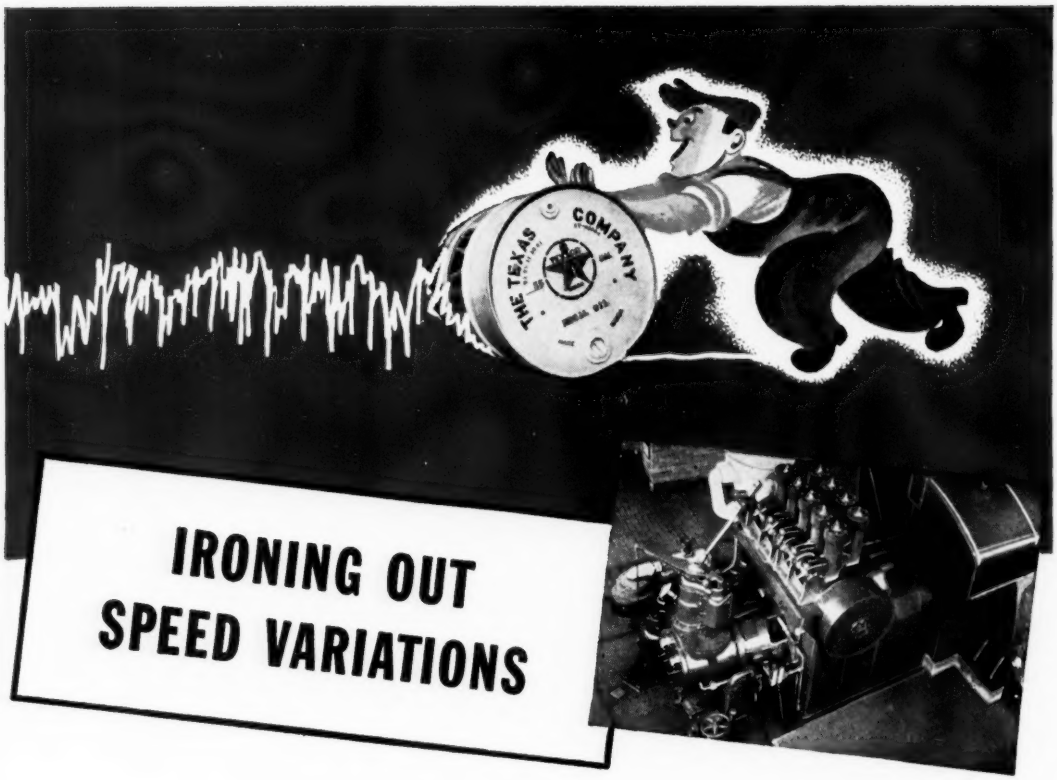

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The Nature of the Lubrication Process

THE development of lubrication theory and practice is a fascinating record of the inquiring nature of the modern scientist. But for this development, it is doubtful that machine design could have progressed as it has, for this progress required continual increase in operating speeds and loads both of which very definitely influence the degree to which lubrication can be maintained.

Obviously, machine design and research into bearing performance required a study of metal structure. The sleeve-type bearing is an alloy ranging from the conventional babbitt combination of copper, tin and antimony, to the more recently perfected cadmium-silver and copper-lead metals. All used to be available in almost unlimited quantities; today, however, national defense requires proration. Tin may become even more restricted.

Bearing protection, therefore, is as much of a patriotic duty as metal conservation. The burden must be borne by lubrication, hence our interest in the nature of the lubrication process. The importance of adequate lubrication results from the fact that machine parts rubbing against one another without adequate lubrication develop so much friction that excessive force is required to move the machine and little is left over to do the work for which the machine is intended, also an inadequately lubricated machine wears out rapidly.

Historians tell us that the need for lubrication was recognized several thousand years ago and they report references to the use of animal, vegetable and mineral oils in very early written records. They report, also, that the mechanized armies of about 1400 B. C. thought very highly of beef and mutton tallow for the lubrication of their chariot wheels.

The Roman naturalist, Pliny (27-79 A.D.) is reported to have made a list of oils which suggests that he recognized the desirability of using heavy oils under some circumstances and light oils under others.

From Pliny to the beginning of the machine age involved a span of some eighteen hundred years—but lubrication theory and practice apparently took it almost in a jump. Of course, there were some experiments and a few things were learned, but on the whole, lubrication in the early machine age was about as crude as in the chariot age. During this period, tallow, olive oil, lard oil, castor oil, whale oil, mineral oils and many others were used in various ways and for various purposes; the selection being based largely on the trial and error principle. Finally, a Russian—N. Petroff by name—found that petroleum oils of appropriate viscosity[1] could be substituted for the other oils then in common use. Petroff found also that the friction of a bearing was dependent on the viscosity of the lubricant.

During this same year Beauchamp Tower, reported to the Institution of Mechanical Engineers[2], the results of friction experiments made on bearings similar to those used in railway cars. The effect on friction of the method of supplying oil was of particular interest in these experiments; therefore, a large number of different arrangements for oil supply was tried. One method, that of immersing the

[1]Viscosity, as will be shown in detail later, is the resistance offered by a fluid to change of shape or relative motion of its parts. It is what molasses has a great deal of in January.

[2]First Report on Friction Experiments—Journal of the Institution of Mechanical Engineers, 1883.

lower part of the shaft in a bath of oil, was found to produce friction many times lower than with the other methods then in common use for the lubrication of machinery.

Toward the end of the experiments with bath lubrication, Mr. Tower made a further important discovery. A hole had been drilled through the center of the bearing at the top to accommodate a lubricator in some future experiments and when the machine was started, still using bath lubrication, oil began to flow out of this hole. The machine was stopped and a wooden plug driven into the hole, but when the machine was re-started the plug was forced out of the hole and again oil flowed out.

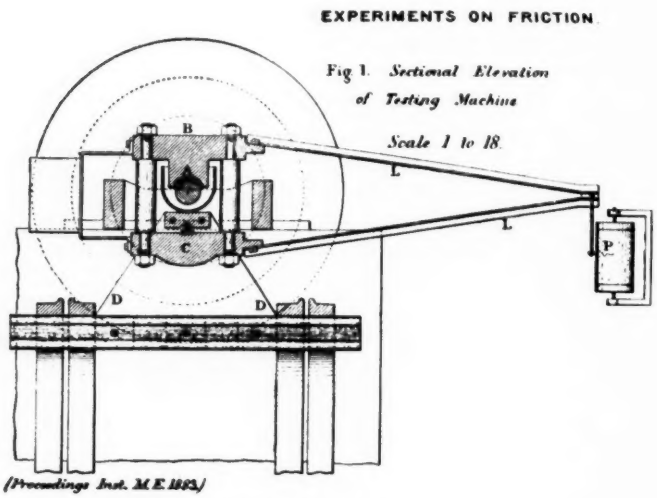

Fig. 1—The Original Tower Testing Machine.

This appears to have been a critical moment in regard to our knowledge of how a lubricant functions: if Tower had successfully plugged this hole our understanding of lubrication probably would have been retarded for many years, but his first attempt to plug the hole failed and he then screwed a pressure gage into the hole and restarted the test. Very shortly after restarting the test with the gage in place, the pressure was so great as to exceed the range of the gage which was capable of measuring pressures only to 200 lbs. per square inch. This experiment gave the first evidence that pressure was created in the oil film of a bearing.

In subsequent experiments[1] employing bath lubrication, Tower drilled several holes in the bearing. On applying pressure gages of suitable measuring capacity to them, he found that the pressure at each hole was different, being zero at the edges of the bearing and greatest at the center. The pressure observed at the center hole was found to be greater than the unit load on the bearing, 625 lbs. per square inch in one case when the unit bearing load was only 330 lbs. per square inch and the pressure increased or decreased when the load on the bearing was increased or decreased. The resultant of the sum of the pressures, measured at the various holes over the bearing surface, was found to equal the load on the bearing.

Figures 1 and 2 are reproductions of illustrations from Tower's reports to the Institution of Mechanical Engineers. Figure 1 accompanied his first report and showed the test apparatus that he used. Figure 2 accompanied the second report and shows how the holes for the pressure measurements were located and gives some of the pressures observed.

Osborne Reynolds[2] was greatly interested in Tower's experiments and in 1886[3] he published a paper based thereon. In this paper Reynolds undertook a theoretical study of the manner in which oil could flow in a space between a bearing and a shaft. As a result of this study

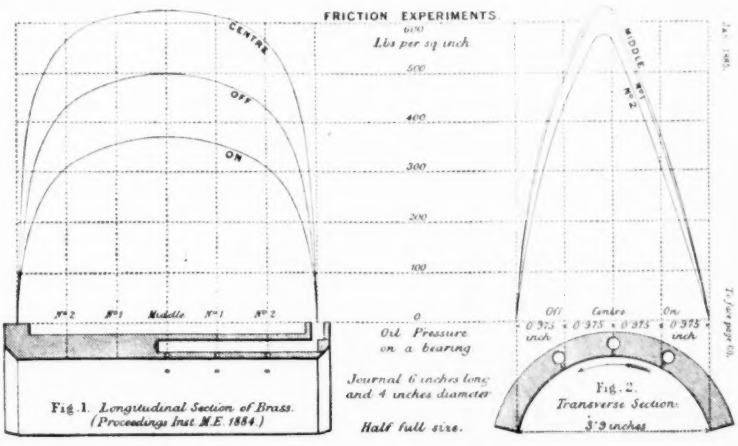

Fig. 2—Data recorded in Tower's friction experiments.

[1]Second Report on Friction Experiments—Journal of Institution of Mechanical Engineers, 1885.

[2]The same Reynolds who is widely known for the Reynolds number which plays such an important part in hydro- and aero-dynamics.

[3]This paper entitled "On The Theory of Lubrication and Its Application to Mr. Beauchamp Tower's Experiments, Including an Experimental Determination of the Viscosity of Olive Oil" was published in the Philosophical Transactions of the Royal Society in 1886.

he developed a description of the lubrication process in which the lubricant was considered to be drawn into a convergent passage between the bearing and the journal where, owing to its viscosity, the lubricant produces forces which account for the load carrying capacity and friction of the bearing. This description is the accepted theory of how a bearing functions and, according to it, viscosity is the essential characteristic of a lubricant that accounts for its being a lubricant.

VISCOSITY

Viscosity is defined as the internal frictional resistance offered by a fluid to change of shape or relative motion of its parts. This property of viscosity appears to have been recognized by the great naturalist, Isaac Newton[1] at about 1687. As in the case of the gravitational laws, also credited to Newton, others had recognized previously the effects of viscosity and realized that fluids exhibited resistance to motion, but Newton was the first to record a useful description of viscosity as a specific property of a lubricant.

The nature of the forces involved in the action of viscosity are conveniently illustrated by an experiment with a deck of cards—see Figure 3. Lay a deck on the table, place a finger on the top of it and push sideways. Unless some of the cards are stickier than others the whole deck will be deformed uniformly as shown—the bottom card remaining stationary and the top card moving with the finger. Each

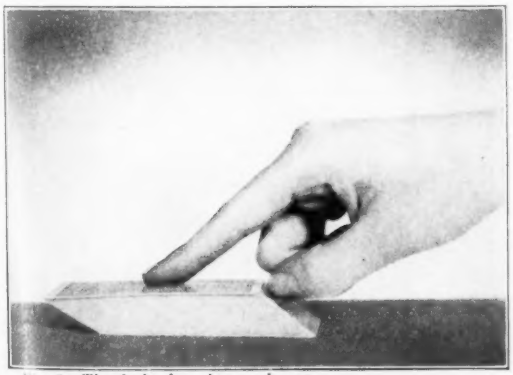

Fig. 3—The deck of cards experiment.

intermediate card presents some resistance to the push and each card slides a little ways. Clearly the resistance to motion of the top card was transmitted uniformly through the deck, the resistance between each pair of cards being equal. The transmission of the push between the successive cards illustrates the transmission of viscous resistance through a fluid; a force of this sort is commonly referred to as a shearing force.

The deck of cards illustrates Newton's idea of the transmission of force from layer to layer

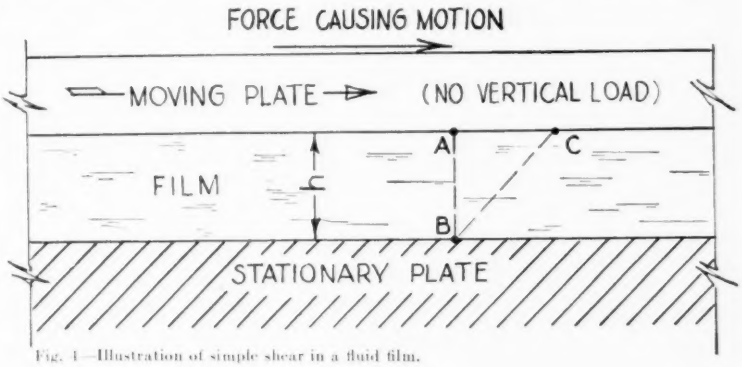

Fig. 4—Illustration of simple shear in a fluid film.

in a viscous fluid but differs in other important respects from what occurs in a viscous fluid. To account for these points of difference, let us consider two plane parallel surfaces with the space between them filled by a viscous liquid, the bottom surface fixed and the top one free to move as in Figure 4.

Practically all viscous liquids suitable for use as lubricants "wet" the surfaces that they contact. With reference to Figure 4 we may consider this "wetting" to mean that fluid particles in contact with the bottom surface will remain fixed to this surface and that fluid particles in contact with the top surface will remain fixed to it. With this idea in mind, if the free upper surface of Figure 4 is considered to have moved so that the point A was moved to the position C, then the fluid particles attached at A will have moved to the new position, C, but the particles at B will have remained at B and the intermediate fluid particles that initially lay along the line A-B, will have been drawn by the viscous forces into the new position C-B, just as the deck of cards deformed.

If two such surfaces are set up and arranged for an experiment so that the force required to move the top surface can be measured, some interesting things will be observed, viz.:

1. Any force, no matter how small will cause the top surface to move relative to the bottom. That is, the fluid cannot support a shearing force except as the result of the viscous forces. Motion must exist to create those forces.
2. If a one-pound force is required to maintain a definite velocity of the upper surface, such as two inches per second, then

[1]The same Newton upon whose head an apple fell and who is known as the discoverer of the Law of Gravitation.

the force required to produce twice the velocity (four inches per second), will be found to be two pounds, and so on. That is, the force varies directly as the velocity.

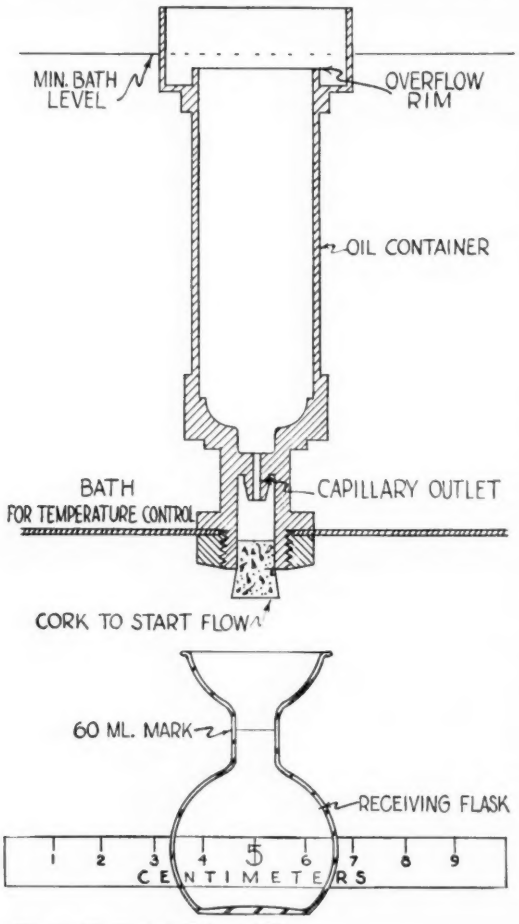

Fig. 5—The Saybolt Universal Viscometer.

3. If we maintain a constant velocity between the surfaces, and alter the separation of the surfaces h, then we will find that when h is cut in half the force will be doubled, or if h is cut to one-quarter of its original value, the force will be increased by four times. That is, the force is inversely proportional to the separation of the surfaces.

This experiment illustrates the ideas involved in Newton's definition of viscosity, but Newton, being a mathematician, wrote these ideas in the form of an equation

$$\frac{F}{A} = Constant \times \frac{V}{h} \quad \ldots\ldots\ldots\ldots 1.$$

or: $\frac{Force}{Area} = Constant\ times\ \frac{Velocity}{Film\ \ Thickness}$

When the velocity is zero, the shearing stress is also zero, which agrees with the observation that a fluid cannot withstand a shearing stress without motion. Further, if h is reduced by half, the value for the force would be doubled, etc., so that the equation represents the observations of the experiment. If the experiment were repeated with some other fluid between the surfaces Equation 1 would still represent the results, but the numerical value of the constant would be different than in the first case. This constant of the equation is the quantity that Newton defined as viscosity. Viscosity is generally designated by the symbol μ (the Greek letter mu), substituting this symbol for the constant of Equation 1 and solving for μ we get

$$\mu = \frac{F}{A} \times \frac{h}{V} \quad \ldots\ldots\ldots\ldots 2.$$

or: $Viscosity = \frac{Force \times Separation\ of\ Surfaces}{Area\ of\ Surfaces \times Relative\ Velocity\ of\ Surfaces}$

Suppose, in an experiment with the two surfaces of Figure 4, we had measured the force as two pounds, the area of the surfaces as ten square inches, the separation of the surfaces as 0.001 inches and the velocity as ten inches per second. By substituting these values into Equation 2, we find that the viscosity

$$\mu = \frac{2\ lbs.}{10\ sq.\ in.} \times \frac{0.001\ in.}{10\ in.\ per\ sec.} = 0.000002\ lbs.\ sec./sq.\ in.$$

This unit of viscosity is called the Reyn in honor of Osborne Reynolds. Another unit known as the Poise is generally used instead of the Reyn. The Poise is defined exactly as the Reyn by Equation 2 but it is designated by the symbol Z, and is measured in units of centimeters, dynes, and seconds instead of inches, pounds and seconds[1]. In this system of units the results of our experiment would have been force = 890,000 dynes, area = 64.52 square centimeters, separation of the surfaces —0.00254 centimeters. Substituting these values into Equation 2, in which we now designate the viscosity by Z instead of μ, the result is

$$Z = \frac{890{,}000\ dynes}{64.52\ sq.\ cm.} \times \frac{0.00254\ cm.}{25.40\ cm./sec.} = 1.38\ dyne\ sec./sq.\ cm.$$

Thus Z differs from μ only by the units in which the measurements are made and if we divide Z by μ, from our experiment we observe that

$$Z/\mu = 69{,}000.$$

So that we can convert from Z to μ or from μ to Z at any time by dividing or multiplying by 69,000, which is simply the difference in the size of the two systems of units.

In actual practice it is very difficult to carry out an experiment such as the one we have been considering, with plane parallel surfaces with sufficient precision for commercial viscosity measurements. For this reason a variety

[1]One inch equals 2.54 centimeters. One pound equals 445,000 dynes.

of types of mechanical apparatus, all involving the same basic ideas as the plane parallel surfaces of Figure 4, have been devised to carry out the measurements with greater precision or with greater convenience. In the United States the most commonly used apparatus for measurement of viscosity is the Saybolt Universal Viscometer shown schematically by Figure 5. In this apparatus there is a container for the fluid to be tested. The fluid is permitted to flow out of the reservoir through the small tube called the capillary outlet into the receiving flask. The time required to fill the receiver is measured.

The fact that the time required to fill the receiver in this apparatus can be interpreted as viscosity depends on the details of what occurs in the capillary outlet. Figure 6 shows an enlarged drawing of the capillary. Referring to the sketch at the top of Figure 6, which shows a cylindrical element of the fluid in the capillary, we see that the weight of the fluid in the container above the capillary exerts a push or pressure on the fluid in the capillary forcing it to flow out. Meanwhile the viscous shearing stresses developed by the motion of this flow tend to resist the flow with the result that a slow steady outflow of the fluid is established and fluid velocities and shearing stresses as indicated by the lower sketches of Figure 6 prevail[1]. In this type of apparatus the weight of the fluid above the capillary corresponds to the force required to push the top surfaces of Figure 4; the time for the 60 milliliter quantity of fluid to flow out of the tube and fill the receiver is the measure of velocity of flow, and the diameter of the capillary corresponds to the thickness of the film. Thus all of the elements required by Newton's equation to define viscosity are present.

As a matter of practical convenience in the operation of the Saybolt test the amount of the fluid above the capillary is maintained constant by filling the receiver to the overflow rim. The volume permitted to flow out is maintained constant and the diameter of the capillary is of course fixed by the construction of the instrument. Therefore, it is necessary only to measure the time from when the cork is pulled to start the flow until the receiver is filled to the 60 milliliter mark on its neck; this time, measured in seconds, is referred to as the Saybolt Universal Viscosity (S.U.V.).

The Saybolt Universal Viscosity is proportional to a quantity called the kinematic viscosity, which, in turn, is equal to the viscosity in absolute units as defined by Newton (Equations 1 and 2) divided by the density of the fluid. This complication that the Saybolt Viscosity is proportional to the kinematic and not to the absolute viscosity, arises from the fact that in the Saybolt apparatus the weight of the fluid above the capillary is what causes the flow through the capillary, and some fluids weigh more than others; the density[2] being a measure of the weight. Figure 7 shows the relationship between Saybolt Universal Viscosity expressed in seconds, and absolute viscosity expressed in Poises and Reyns for oils of different density.

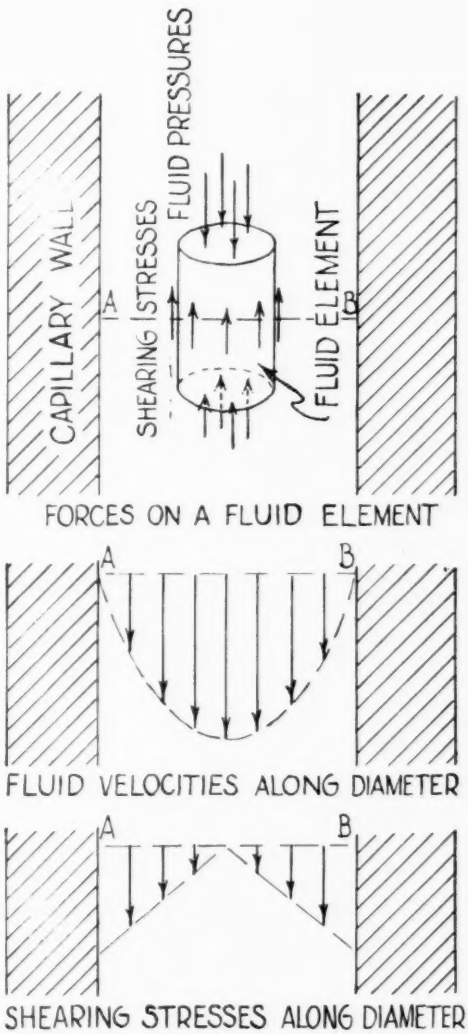

Fig. 6—Enlarged drawing of the capillary showing principles of capillary flow.

[1]In the Saybolt apparatus there are several other factors involved in the interpretation of the time measurement in terms of viscosity. It is true that the viscous forces resist outflow of the fluid in the manner described above; however, the changing head of the fluid and the short length of the capillary lead to complications so that the final interpretation of the result as viscosity is dependent on calibration of the apparatus with fluids of accurately known viscosity as determined by other more fundamental methods of measurement.

[2]In centimeter-dyne-second units density is numerically equal to specific gravity and the latter data are most generally available so it is most commonly used instead of density.

When measurements of the viscosity of a fluid are made at different temperatures, one quickly finds that the viscosity is very greatly dependent on temperature, becoming much less for a given fluid as the temperature increases. It is necessary, therefore, to specify the temperature at which the viscosity is measured, three temperatures in common use being: 100, 130 and 210 degrees Fahr., and the result of a viscosity test is expressed as so many Reyns, Poises or Seconds Saybolt, as the case may be at the test temperature (thus 322 S.U.V. seconds at 130 degrees Fahr.). Figure 8 shows measurements of viscosity at a series of temperatures for two different oils. The very great dependence of viscosity on temperature is clearly evident.

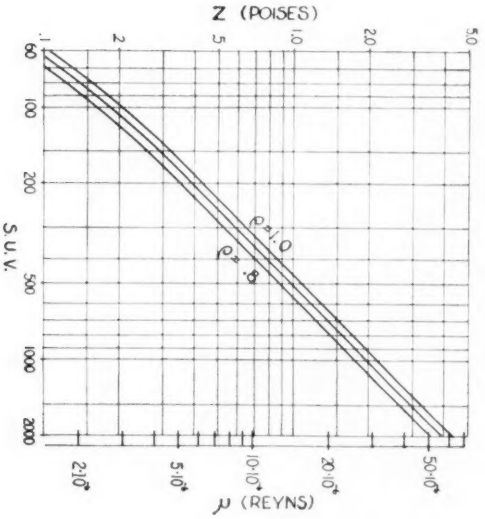

Fig. 7—The symbol ρ designates specific gravity; thus the upper line applies for oils of specific gravity 1.0, the center for oils of specific gravity 0.9, and the lower for oils of specific gravity 0.8. μ and Z are given in the customary units, pounds, seconds per square inch, and dynes, seconds per square centimeter.

In the study of any lubrication problem it is necessary to know the viscosity of the lubricant at the temperature at which it will be used. To avoid having to make measurements at a large number of temperatures, diagrams have been devised from which viscosity at any temperature can be estimated with a fair degree of accuracy from observations at any two temperatures. Figure 9 shows the current ASTM Viscosity-Temperature Chart. The original idea was first developed by The Texas Company and published in LUBRICATION in June, 1921. Since then it has been revised and adopted by the ASTM and is in current use by the industry.[1] The lines drawn on this chart represent the viscosity data that were shown in Figure 8; it is clear that when viscosity is plotted against temperature on this chart, the scale is so constructed as to cause the results to lie on a straight line. Thus, to determine the viscosity of an oil at any temperature, it is possible to plot on this chart observations made at two temperatures and draw through them a straight line from which the viscosity at other temperatures can be determined.[2] It is important to remember that this chart is applicable only in the viscous flow region, i.e., down to around 40 seconds viscosity. Below this, in the turbulent flow region, it does not apply.

THE ACTION OF A VISCOUS FLUID BETWEEN MOVING SURFACES

With the property of viscosity and the nature of the viscous forces in mind, we now can return to a study of the action of a lubricant in a bearing in order to determine how its action is related to the load carrying capacity and frictional resistance of bearings. This is described in terms of three simple and separate cases which finally can be combined to represent the action of a lubricant in a bearing.

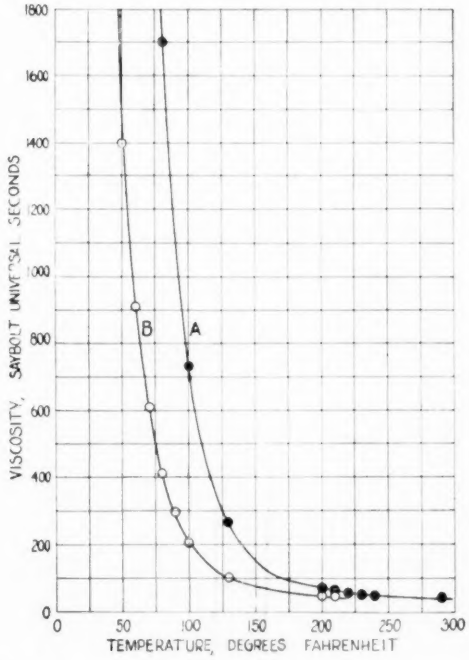

Fig. 8—Temperature-Viscosity relationship for two straight run distillates.

[1] See Viscosity-Temperature Charts for liquid petroleum products (D341-39) ASTM Standards III P. 221.

[2] Many oils tend to solidify at low temperatures and under such conditions they cannot be considered viscous fluids to which Fig. 9 applies. The pour point temperature which is the least temperature at which an oil will pour can be used as a guide to how far Figure 9 applies and in any particular case Figure 9 should be used only to a low temperature that is a few degrees greater than the pour point temperature.

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The description follows closely the one given by Osborne Reynolds in his original paper on the theory of lubrication.

Case I. Parallel Surfaces—One Sliding Across the Other

In Figure 10, the surface AB is supposedly fixed while the surface CD moves to the right

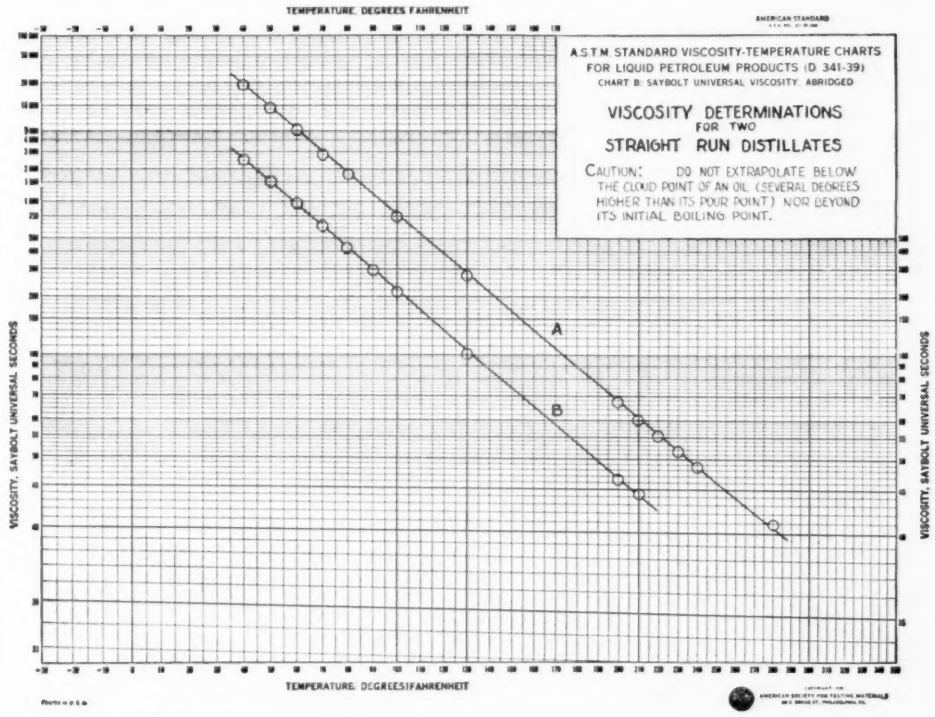
TEMPERATURE, DEGREES FAHRENHEIT
A.S.T.M. STANDARD VISCOSITY-TEMPERATURE CHARTS
FOR LIQUID PETROLEUM PRODUCTS (D 341-39)
CHART B: SAYBOLT UNIVERSAL VISCOSITY. ABRIDGED
VISCOSITY DETERMINATIONS
FOR TWO
STRAIGHT RUN DISTILLATES
CAUTION: DO NOT EXTRAPOLATE BELOW THE CLOUD POINT OF AN OIL (SEVERAL DEGREES HIGHER THAN ITS POUR POINT) NOR BEYOND ITS INITIAL BOILING POINT.
A
B
VISCOSITY, SAYBOLT UNIVERSAL SECONDS
VISCOSITY, SAYBOLT UNIVERSAL SECONDS
TEMPERATURE, DEGREES FAHRENHEIT

Fig. 9—The A.S.T.M. Viscosity-Temperature Chart.

with velocity V and the surfaces are separated by the distance h. If the point E moves to the position F (also point G to position H), the fluid particles along AE move to the line AF and additional fluid follows these particles into the space separating the surfaces. The triangle $AEFA$ represents the quantity of oil that has flowed into the clearance space across the entrance AE. If EF was drawn so that it represents the distance in inches moved by the surface CD in one second, then EF is equal to the velocity in inches per second and the area of the triangle $AEFA$ becomes

$$Area = \frac{V \times h}{2}$$

which is also equal to the volume[1] of fluid that flows into the clearance space in one second. The quantity flowing out from the surface at BG will also be equal to $\frac{Vh}{2}$.

It is essential that the quantity flowing out of the clearance space be equal to that flowing in, because most lubricants are practically incompressible and so we cannot cram into the clearance space a volume of lubricant that is greater than the volume of the clearance space.

Case II. Parallel Surfaces Approaching with no Sliding

In Figure 11, the surface AB is again supposedly fixed while the surface CD moves toward AB and the space between the surfaces is filled with the viscous fluid. As CD approaches AB, the fluid evidently must be squeezed out from between the surfaces at the edges AE and BG. Because the fluid "wets" the surface, fluid particles at the surfaces will remain fixed to the surfaces and the outflow will be a maximum midway between them.

If, at a certain point in the motion, as shown by the dotted lines of Figure 11, the clearance space is divided into equal parts by dotted lines and these lines are supposed to move with the fluid, they will, after a short interval of the motion of CD assume the position of the solid lines of Figure 11. Thus, the fluid initially in the space $AEFHJA$ will, after the motion, be in position $AIEFJHJA$. The area $AIEA$ will represent the outflow from the clearance space passing the edge AE. The total outflow

[1]Per unit length of the surface measured in the direction perpendicular to the page.

from the clearance space will be the sum of the areas $AIEA + BKGB$ and must equal the decrease in the area between the surfaces resulting from the motion of the plate CD from its initial dotted position to its final position as shown by the solid lines of the figure.

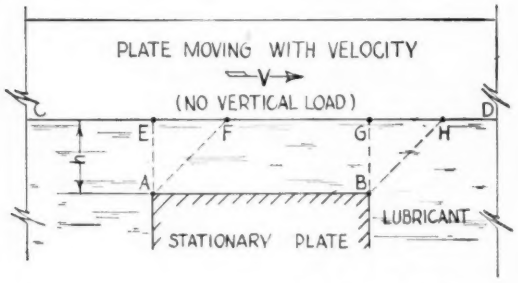

Fig. 10—Illustration of sliding parallel surfaces. See Case I.

The outflow of the fluid from between the surfaces will be resisted by the viscosity of the fluid much as though the lines such as AIE were elastics that have to be stretched by the fluid as it flows out of the clearance. This action will cause a pressure to be developed in the space between the surfaces and since the edges of the surfaces at A and B are at atmospheric pressure, the pressure between the surfaces will be distributed substantially as indicated on Figure 11.

The distribution of velocity and shearing stresses in the fluid during the outflow from the clearance space will be practically identical with the distribution shown by Figure 6. In the present case as in the capillary tube of the Saybolt Viscometer, Figure 6, the flow was created by pressure.

Case III. Inclined Surface, One Sliding Across the Other

This case, Figure 12, is identical with Case I except that the stationary surface AB is inclined so that the separation at the inlet AE is greater than at the outlet BG. As in Case I, the area $AEFA$ represents the quantity of fluid that the motion of CD attempts to draw into the space between the surfaces, and $BGHB$ represents the quantity that the motion of CD attempts to carry away at BG, both being the quantities resulting from the motion of CD relative to AB only.

In this case the inflow at AE is in excess of the outflow at BG (IEFI = BGHB) which is impossible so that something else must happen. What happens is that the viscous forces attempt to wedge more fluid into the clearance space than can get out and, in consequence, develop a pressure in the film with a distribution as indicated in Figure 13. This pressure decreases the inflow and aids in creating a larger outflow with the result shown by Figure 13 that the inflow is decreased to a new area $AEFIA$ and the outflow is increased to a larger area $BGHJB$ ($AEFIA = BGHJB$). The area $AEFIA$ is equal to the area $AEFA$ of Figure 12, minus the area $AFIA$ which represents outflow tendency at AE resulting from the pressure in the film as in Case II. Similarly the outflow at BG is equal to the area $BGHB$ of Figure 12 plus the area $BHJB$ resulting from the pressure in the film as in Case II.

The situation presented by Figure 13 involves a combination of the actions represented separately by Figures 10, 11 and 12, and it represents the condition that Osborne Reynolds considered to be necessary for fluid film lubrication; namely, a continuously available supply of a viscous fluid drawn by relative motion of the bearing surfaces into a wedge-shaped clearance space between the surfaces.

In Figure 13, the motion of the surface CD will be resisted by forces arising from the viscosity of the lubricant and these forces are the source of the friction of the bearing. These same viscous forces also produce a pressure in the oil film. It is clear that these film pressures will exert a force on the plate AB equal to the product of the average pressure and the area of AB and this is the source of the load carrying capacity of a fluid lubricated bearing.

It is not clear at first that the wedging of the fluid into the space between the surfaces, by the relatively weak viscous forces can lead to high enough pressures to account for the known large carrying capacity of bearings. However, the sketches drawn to show the flow of the fluid were of necessity exaggerated and create

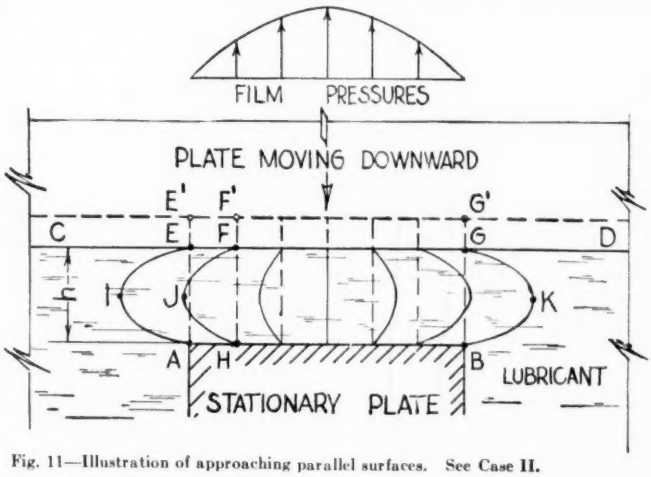

Fig. 11—Illustration of approaching parallel surfaces. See Case II.

a somewhat erroneous idea of the separation and the inclination of the surface. In a practical case, the separation of the surfaces at BG might be 0.0005 inches and the inclination (excess of AE over BG) 0.0002 inches in a length AB of several inches so that the effect is that of a very flat wedge giving a tremendous mechanical advantage and the reasonableness of large pressures being produced is much more apparent.

The foregoing three cases have described the nature of the lubrication process in terms of flat surfaces which, though important, are not encountered in lubrication practice as frequently as cylindrical surface bearings. Fortunately, it is possible to apply the ideas developed in connection with flat surfaces directly to cylindrical bearings. As shown in Figure 14, for a cylindrical bearing, the moving surface CD has been wrapped around into a circle so that it becomes the shafting and the stationary surface AB has been wrapped around so that it becomes the bearing.

In the cylindrical bearing the ideas concerning oil flow, film pressure, and frictional resistance developed in connection with the inclined surfaces of Figure 13 can be applied almost without modification. The wedge-shaped passage between the surfaces results from the fact that the bearing is commonly made to have a few thousandths greater diameter than the shaft, and the shaft runs with its center displaced from the bearing center, as indicated by Figure 14. Oil flow into and out of the space between the surfaces at the entrance, AE, and at the exit, BG, is represented in the figure by the areas, AEF and BGH, which have the

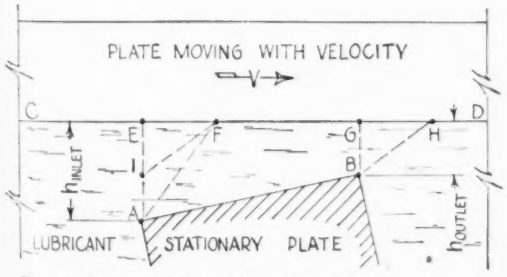

Fig. 12—Inclined sliding surfaces. See Case III.

same significance as in the plane surface case of Figure 13. Action of the viscous forces causes pressure to be developed in the clearance space and the resultant of the pressure supports the load carried by the shaft.

THE PERFORMANCE OF BEARINGS

The nature of the lubrication process outlined by the foregoing three cases and amplified by our knowledge of the forces of viscosity leads to an understanding of why and how an oil lubricates a bearing. It is necessary further

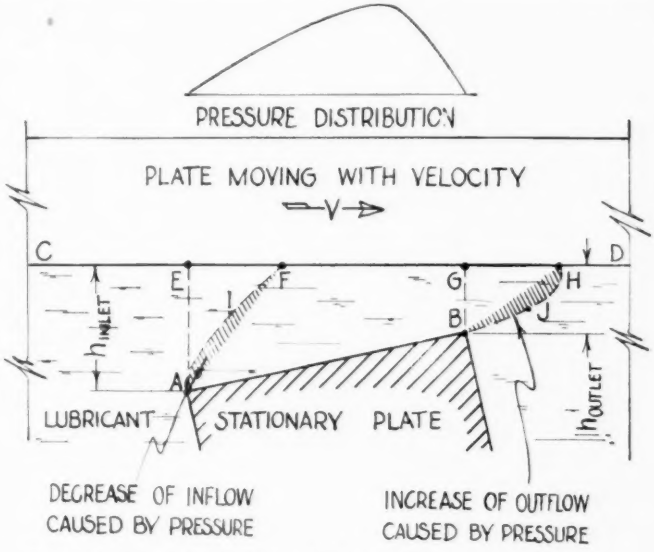

Fig. 13—Inclined sliding surfaces. See Case III.

to interpret this description of the lubrication process in terms of the practical performance characteristics of bearings.

From Newton's definition of viscosity it is clear that the viscous forces resisting motion depend on the viscosity μ, the velocity V and the separation of the surfaces h. If the separation of the surfaces is kept constant then the viscous forces are proportional to the product of viscosity and velocity $\mu \times V$. If the load on the bearing increases, with $\mu \times V$ constant, some increase in the viscous forces must occur to support the increased load and this results from a decrease of the film thickness h. Thus the thickness of the film is dependent on the load W[1].

These ideas enable us to write down a quantity $\frac{\mu V}{W}$. It turns out, both from mathematical studies of lubrication and from the examination of experimental data, that this quantity can be regarded as a single number that describes the conditions in the oil film. For example, in an experiment the performance of a bearing will be found to depend on viscosity, velocity and load. To represent the results of this experiment, three groups of graphs will be required showing respectively

[1]W represents the load in pounds per inch of surface measured perpendicular to the page as on Figure 13.

the dependence of performance on viscosity, velocity and load. The number $\frac{\mu V}{W}$ describes the conditions in the oil film in such a way that if performance is plotted against $\frac{\mu V}{W}$ all of these graphs will collapse into a single line. To be somewhat more specific a given value of $\frac{\mu V}{W}$, say 0.001 could result from $\mu = 0.001$, V = 100, W = 100 or from $\mu = 0.01$, V 1000 W = 10,000 or from any other combinations of μ, V and W which lead to the value $\frac{\mu V}{W} = 0.001$, and the performance of a fluid film lubricated bearing under all of these conditions is the same so long as the combination $\frac{\mu V}{W}$ continues to equal 0.001.

In connection with the lubrication of cylindrical bearings it has become common practice to write instead of $\frac{\mu V}{W}$ a term $\frac{ZN}{P}$ which is derived directly from $\frac{\mu V}{W}$ as follows:

Z is the viscosity in Poises and equals $\mu \times$ 69,000, or, Z is sometimes written as Centipoises: 1 Centipoise equals 1/100 Poises.

N is the shaft speed measured in RPM. Note that the velocity V equals $\pi D N$, where: π has its usual significance 3.1416, D is the shaft diameter and for a given bearing is a constant, so that N is the only variable quantity.

P is the load on the bearing, usually in pounds per square inch of projected bearing area. This is the total load on the bearing divided by the projected area which is usually figured as the product of the diameter and the axial length.

The quantity $\frac{ZN}{P}$ like $\frac{\mu V}{W}$ from which it originates may be regarded as a single number that describes the condition of the oil film[1].

Friction and load carrying capacity of the bearing are of predominant importance as measures of bearing performance. Friction determines the power loss in the bearing and it is generally desirable that it be as small as possible. Load carrying capacity of the bearing is dependent on the ability of the film to keep the surfaces separated and avoid metal-to-metal contact. Therefore, separation of the surfaces becomes a measure of the load carrying capacity and of the safety of the bearing —the greater the separation of the surfaces the less the likelihood of failure due to metal-to-metal contact or to the destructive action of foreign particles in the lubricant. The friction and separation of the surfaces in a bearing are related to the number $\frac{ZN}{P}$.

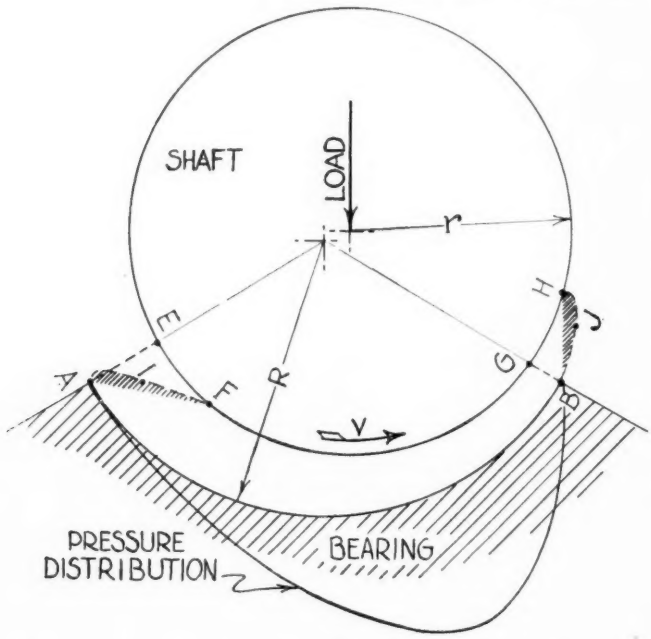

Fig. 14—A 120° cylindrical bearing. R is the bearing radius, usually several thousandths of an inch greater than the shaft radius r.

Friction in a bearing is usually given in terms of the coefficient of friction which is defined as the quotient of friction force F divided by the total load on the surface W; usually it is designated by the symbol f (the Greek letter lambda, λ is also used on some occasions).

Separation of the bearing surfaces is conveniently given in terms of the separation at the point of closest approach, called the minimum oil film thickness and designated by the symbol h_o. Metal-to-metal contact or other disruption of the oil film is most likely to occur at this point.

Figure 15 shows the results of measurements of coefficient of friction and minimum film thickness plotted against $\frac{ZN}{P}$ for a bearing completely surrounding the shaft. The test

[1]In actual use of the number $\frac{ZN}{P}$, the viscosity Z is sometimes given in Poises and sometimes in Centipoises, and the unit load P is sometimes figured in different ways so that in any practical case it is necessary to investigate the definition of these quantities to make sure just how they are defined in that case.

bearing was 2½ in. in diameter, 3⅞ in. long and had a radial clearance of 0.005 in., test speeds ranged from 500 to 3600 RPM, loads from 100 to 500 lbs. per square inch of projected area and oil viscosities from *19* to *83* Centipoises. These diagrams may be considered to apply quite generally to bearings that are geometrically similar to the test bearing on which the data were determined. The geometrical similarity restriction requires that the bearings have approximately the same ratio of axial length to diameter $\frac{L}{D} = 1.55$ and the same ratio of diametral clearance to diameter, $\frac{C}{D} = 0.002$.

On Figure 15[1] it is important to distinguish the two regions of lubrication—

Fluid Film Lubrication prevails in the region of $\frac{ZN}{P}$ values above 60 to 80. Here the surfaces are separated completely by the film of lubricant and the friction is generally very low and is dependent entirely on the viscous forces.

Boundary Lubrication prevails at values of $\frac{ZN}{P}$ below 60 to 80. Here the surfaces are in partial metal-to-metal contact and the friction may become very great even up to seizure of the surfaces. This region also has been called the "region of unstable lubrication." This name is derived from the fact that friction increases as $\frac{ZN}{P}$ becomes less so that once in this region a bearing will begin to heat up owing to the increased friction. This heating will reduce the viscosity increasing the metal-to-metal contact and causing higher friction which in turn will cause further increase in temperature, further reduction of viscosity and still more increase of friction and so on until some limiting condition, often failure of the bearing by seizing, is reached.

In the foregoing description of the operation of the lubricant it has been assumed that the surfaces of the shaft and of the bearing were geometrically perfect. In actual practice this condition is not realized, and all real bearing surfaces have some irregularities. These irregularities produce an important modification of the theory as they limit the closeness with which the surfaces may approach before metal-to-metal contact occurs. The modification of the theory created by these irregularities makes itself evident in the locations of the transition point from fluid to boundary lubrication, and it is extremely important to recognize that this transition depends not on the value of $\frac{ZN}{P}$ but on the minimum film thickness at which metal-to-metal contact occurs. Thus, with a rougher bearing than was used in the tests from which the data of Figure 15 were derived, the transition would have occurred at a higher value of $\frac{ZN}{P}$, while with a smoother bearing it would have occurred at a lower value of $\frac{ZN}{P}$.

Data of the type given by Figure 15 generally form the basis for design of bearings or for analysis of problems in cases of bearings that do not function properly.

In planing a bearing for operation on a piece of machinery where minimum power loss is of less concern than reliability and operating safety, and where the lubricant supply may at times be contaminated with fine particles of solid material, it would be wise to select a value of $\frac{ZN}{P}$ that would produce a relatively

[1]Based on data developed at Pennsylvania State College in 1932 under a fellowship sponsored by The Texas Company.

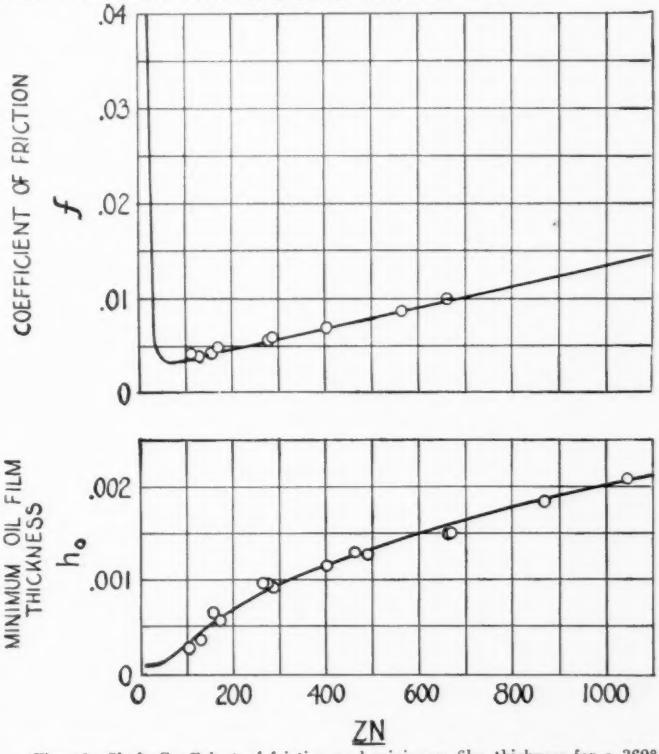

Fig. 15—Shaft Coefficient of friction and minimum film thickness for a 360° bearing. Z is in centipoises; N rev. per min.; and P, pounds per square inch of projected area.

large separation of the surfaces of the shaft and bearing—say for example, $\frac{ZN}{P} = 980$, giving a minimum film thickness of 0.002 in.

The shaft speed N and the total load on the bearing W are presumably fixed by the nature of the machine so that only Z and P can be selected. Z can be made large by the selection of a highly viscous oil, though caution must be exercised in this direction. P can be made small by building the bearing large. Suppose, for example, in the circumstance proposed, that the shaft speed must be 1200 RPM and the total load 900 lbs. The bearing size is selected as 2.32 in. diameter by 4.23 in. axial length giving a projected area of 12 square inches and a unit load, P equals 75 lbs. per square inch. Then to attain $\frac{ZN}{P} = 980$ the lubricant must be selected to have a viscosity of 60 Centipoises (at the operating temperature of the bearing).

In contrast to the foregoing example, minimum friction is important. The lubricant can be handled to assure freedom from foreign particles, and special care can be taken to insure good finishing of the shaft and bearing surfaces. In such a case these extra precautions may make it possible for the bearing to operate safely with a very much reduced minimum film thickness. To take advantage of the correspondingly lower friction loss—we decide that a value of $\frac{ZN}{P}$ of 160, giving a minimum film thickness of 0.0005 in. will be satisfactory. Again the shaft speed is 1200 RPM, the total load to be carried is 900 lbs. and the bearing size is selected to give a projected area of 12 square inches. In this case the oil can be selected to have a viscosity of 10 Centipoises at the operating temperature.

Comparing these two examples we notice that the coefficient of friction in our first design will be 0.013 while in the second it will be only 0.004. Thus, the friction loss of the second bearing is only about 1/3 of that of the first bearing. The temperature rise of the bearing is proportional to the friction so that the second bearing will also run substantially cooler than the first. On the other hand the second bearing is close to the region of unstable lubrication and a relatively small increase of load or decrease of viscosity owing to some unusual circumstance could cause it to operate in that region while the first bearing can stand large disturbances without danger of unstable lubrication. Also the second bearing is much more liable to incur damage from contamination of the lubricant so that it would generally require more elaborate construction details to guard it against such damage.

SUMMARY

To maintain operation of a bearing under the condition of fluid film lubrication there must be an ample supply of lubricant drawn by relative motion of the surfaces into a wedge shaped passage between the surfaces. When these conditions exist, the forces of viscosity produce two important results:

1. The viscous forces produce frictional resistance to the relative motion of the surfaces. This viscous friction is very much less than the friction which results when the surfaces come into metallic contact.
2. The viscous forces wedging the fluid into the passage between the surfaces cause a pressure to be created in the film and this pressure holds the surfaces apart. This of course requires that the sum of the pressures be equal to the load applied to the bearing.

   Since the surfaces are separated by a fluid film no wear can occur.

The fluid film may exist with any viscous fluid; even air, for example, possesses the required properties to such an extent that air lubricated bearings have been made and shown to float on a film of air.

In practical application any one of a large number of viscous fluids could be used as a lubricant. However, mineral oils are most commonly used because they are available in large quantities, can be made to almost any desired viscosity and possess other desirable properties such as oxidation stability which make them peculiarly well suited for use as lubricants.

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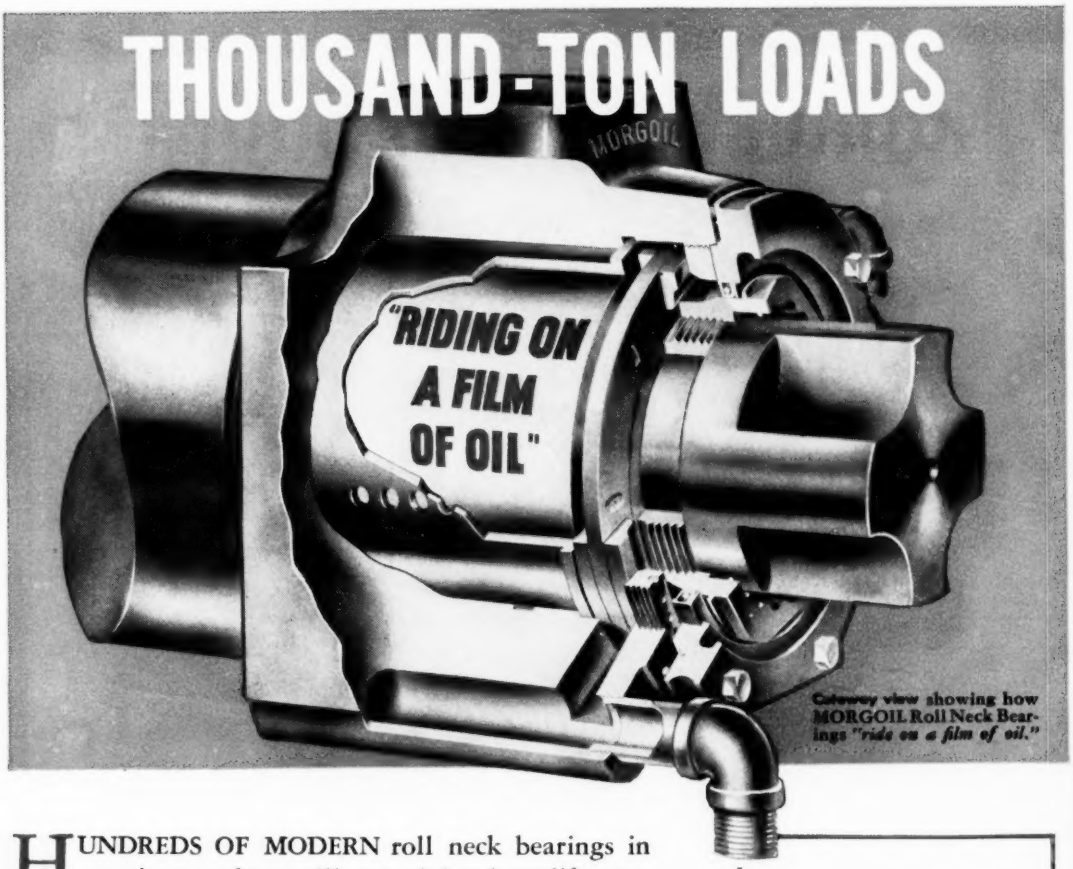

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